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THE FLOW OF A LIQUID-VAPOR MIXTURE THROUGH A DUCT.(U)

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THE FLOW OF A LIQUID-VAPOR MIXTURE THROUGH A DUCT

William P. Walters
John Boyle

August 1977

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USA ARMAMENT RESEARCH AND DEVELOPMENT COMMAND
USA BALLISTIC RESEARCH LABORATORY
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The critical and subcritical mass flow rate is calculated for a homogeneous, adiabatic, frozen, liquid-vapor mixture flowing through a vent with friction. A critical pressure ratio (defined based on the simplifying assumptions) and other fluid dynamic parameters are calculated. The model is an extension of the compressible, adiabatic, constant duct area gas dynamics equations with wall shear. Next page

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In spite of the limiting assumptions, the model is in agreement with experimental data and other critical flow analytical models. The model is capable of predicting a critical pressure ratio, predicting flow rate for critical or subcritical flow, predicting all fluid properties of interest for either critical or subcritical flow, and the model employs a sonic velocity expression consistent with the assumptions.

This model has been successfully employed to aid in predicting the pressure transients of nuclear reactor subcompartments following a postulated loss-of-coolant accident.

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I. INTRODUCTION

The flow of liquid-vapor mixtures occurs in many engineering applications, such as drains, the handling of refrigerants and condensed gases, blow-offs from turbines or boilers, spacecraft propulsion and attitude control systems, and in the coolant system design for nuclear power plants.

To accurately analyze a two-phase flow it would be advantageous to know, a priori, whether the flow is critical or subcritical. The ability to predict both critical and subcritical flow rates, as well as other dynamic and thermodynamic properties of the flow field, is important to the understanding of these problems.

Two-phase flow has been treated extensively in the literature, due primarily to the advances in nuclear reactor technology. The emphasis of the two-phase flow research, however, has been to predict the critical flow rate and the pressure wave propagation. Compressible, subcritical two-phase flow has received relatively little attention, probably because maximum or critical discharge rates resulting from ruptured steam lines are of primary importance in the design of nuclear power plant coolant systems. Subsonic flow is usually treated as incompressible. The Bibliography bears witness to these facts.

Two-phase flow analyses are complicated by departures from equilibrium (metastability), slip between the two phases (non-homogeneity), and an undefined sonic or critical flow velocity concept. Also, two-phase flow analyses usually depend on the quality range, void fraction range, and the flow regimes under investigation. Multi-component, two-phase flow analyses introduce additional complications. A complete description of the two-phase flow phenomenon should include an accurate subsonic, compressible flow model; a critical pressure ratio or other criterion to describe the onset of critical flow; a critical, compressible flow model; and a pressure wave propagation prediction compatible with the subsonic and critical flow models. These models should be dependent on the flow regimes defined by the void fractions, slip ratios, mass flow rates, and qualities associated with these flows.

The existing two-phase flow models are primarily designed to predict the critical flow rate of a steam-water mixture discharging from a nozzle, short pipe, or orifice [1-27]. The sonic velocity or pressure wave propagation in a two-phase flow has been studied in [3, 27-48, 14-17, 21, 23, 24]. Items [7, 46, 47] are recent survey articles supplemented by the Bibliography given here.

To date, the models proposed by Fauske or Henry [5, 7, 8, 9, 11, 12, 18, 25, 26, 31, 49] or those proposed by Moody [1-3] are the most popular methods for predicting two-phase, one-component (steam-water) critical flow rates. The Aisch model [22] is capable of predicting either critical or subcritical flow, and liquid effects are included through an isentropic specific heat ratio and in the density. Large scale computer codes, such as RELAP4 [50] calculate either subcritical or critical flow, but this is accomplished by taking the lesser of several calculated flow rates, for a conservative (low) estimate of the flow rate. Thus, the onset of critical flow is not predicted.

II. THE VENT FLOW MODEL

A two-phase, single-component, homogeneous, frozen, adiabatic vent flow model is presented. This model is applicable to problems associated with the postulated rupture of high energy pipe lines in nuclear power plant coolant systems and the resulting subcompartment pressure buildups. Items [51 and 52] of the Bibliography provide a detailed description of the overall problem. This analysis considers, in effect, only one component. Multi-component (several vapors or gases) can be incorporated into the analysis by appropriately weighting the gas components by standard methods such as discussed by [22].

At any given time during the transient, the total conditions in the nodes upstream and downstream of the vent are assumed given, see [52], for example. Then, the subcritical or critical flow that will occur across the vent, with a specified area and resistance coefficient, as a result of the imposed pressure differential can be determined. This mass flow rate is then used to update the node inventories, and the procedure is repeated.

The vent flow model considers two cases. First, isentropic inlet effects are included in the flow model when the vent is best represented by an area reduction (contraction). Second, when a contraction of the flow does not occur, such as in subcompartment nodalization studies, the isentropic inlet effects are not included.

At the onset, the following assumptions are made:

1. The flow is quasi-steady, i.e., the flow at any point in time is assumed steady.

2. The flow is one-dimensional.

3. The flow is homogeneous.

4. The flow is adiabatic.

5. No mass transfer occurs between phases.

6. The vapor phase of the mixture is a thermally and calorically perfect gas.

7. Pressure changes within the vent due to gravity are negligible.

8. The vent loss coefficient is constant. However, a two-phase flow multiplier could be employed, or any two-phase friction coefficient model which can be represented in terms of the unknown flow field parameters could be used. Thus, this assumption may be removed if a two-phase flow loss coefficient expression is known. Also, *vena contracta* losses are included in the vent loss coefficient.

9. The vent area is constant during a time step.

10. The liquid phase is incompressible over the specified range of pressures and qualities.

A geometric representation of the flow field under discussion is depicted in Figure 1.

The continuity, momentum, energy and state equations are solved for a two-phase, single-component mixture using the assumptions given earlier. Algebraic equations are generated which allow the determination of the vent inlet and outlet Mach numbers based on knowledge of the total pressure ratio across the duct and the vent friction factor. Knowledge of the inlet and exit Mach numbers in turn allows the calculation of other flow field parameters. This model also enables critical flows to be calculated based on the criterion that critical flow coincides with sonic flow at the duct exit.

The governing equations follow.

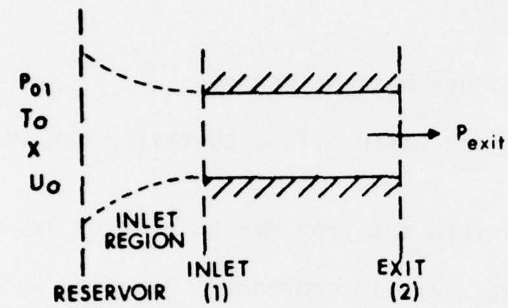

Figure 1. Two-Phase Flow Through A Constant Area Duct

The stagnation enthalpy is conserved, i.e.,

$$dh + \frac{du^2}{2} = dh_o = 0, \tag{1}$$

where

$$dh = Xdh_g + (1-X)\, dh_f.$$

The vapor phase is an ideal gas, so $dh_g = C_p dT_g$.

The total derivative of the liquid phase enthalpy is

$$dh_f = \left(\frac{\partial h_f}{\partial T_f}\right) dT_f + \left(\frac{\partial h_f}{\partial v_f}\right) dv_f = 0,$$

because the liquid is incompressible and adiabatic flow is assumed.

The flow is assumed homogeneous, so the energy equation (1) becomes

$$XC_{p_g} dT_g + \frac{du^2}{2} = 0, \tag{2}$$

where the quality, X, is defined by

$$X = \frac{m_g}{m_g + m_f}.$$

The specific volume is given by

$$v = Xv_g + (1-X)v_f$$

or

$$\frac{v}{v_g} = X, \tag{3}$$

for $v_g >> v_f$. In fact, this approximation is acceptable (to within 5%) for qualities above 0.2 and pressures below 2070 k Pa.

The equation of state for an ideal gas is

$$Pv_g = RT$$

or

$$Pv = XRT. \tag{4}$$

The sonic velocity of a homogeneous, frozen two-phase mixture, neglecting the compressibility of the liquid is [31]

$$\frac{a_g^2}{a_{TP}^2} = \alpha^2 \left(1 - \rho_f/\rho_g\right) + \alpha\,(\rho_f/\rho_g). \tag{5}$$

Introducing the void fraction, α, of a homogeneous two-phase flow mixture,

$$\alpha = \frac{X\,\rho_f}{(1-X)\rho_g + X\rho_f} \tag{6}$$

the sonic velocity becomes,

$$a_{TP}^2 = X\,a_g^{\,2} \quad \text{for} \quad \rho_g << \rho_f. \tag{7}$$

The two-phase sonic velocity for an ideal gas becomes

$$a_{TP}^2 = X\gamma RT_g = \gamma Pv. \tag{8}$$

Thus, the homogeneous two-phase Mach number may be expressed as

$$M^2 = \frac{u^2}{X\gamma RT} = \frac{u^2\rho}{\gamma P}. \quad (9)$$

Thus, equation (2) becomes, for $C_p = \frac{\gamma R}{\gamma - 1}$,

$$\frac{dT}{T} + \frac{\gamma-1}{2} M^2 \frac{du^2}{u^2} = 0. \quad (10)$$

For the flow through a duct, the momentum equation may be written

$$\rho u du + dP + \frac{\rho u^2}{2} \frac{fdL}{D} = 0, \quad (11)$$

where f is the resistance coefficient (or four times the friction coefficient defined by Shapiro [53]).

The continuity equation is

$$G = \rho u,$$

or

$$\frac{1}{2} \frac{du^2}{u^2} + \frac{d\rho}{\rho} = 0. \quad (12)$$

Using the definition of Mach number, (9), the momentum equation (11) may be rewritten as

$$\frac{\gamma M^2}{2} \frac{du^2}{u^2} + \frac{dP}{P} + \frac{\gamma}{2} M^2 \frac{fdL}{D} = 0. \quad (13)$$

The equation of state, (4) may be expressed as

$$\frac{dP}{P} = \frac{d\rho}{\rho} + \frac{dT}{T} \quad (14)$$

and (10) becomes

$$\frac{dP}{P} - \frac{d\rho}{\rho} + \frac{\gamma-1}{2} M^2 \frac{du^2}{u^2} = 0. \qquad (15)$$

Equations (12) and (15) may be combined to yield

$$\frac{du^2}{u^2} = \frac{-\ dP/P}{1/2[1+(\gamma-1)M^2]} . \qquad (16)$$

Thus, (13) may be written as

$$\frac{dP}{P} = \frac{-\gamma M^2[1+(\gamma-1)M^2]}{2[1-M^2]} \frac{fdL}{D} . \qquad (17)$$

Now returning to (9) and using logarithmic differentiation,

$$\frac{dM^2}{M^2} = \frac{du^2}{u^2} - \frac{dT}{T} \qquad (18)$$

or $\frac{dM^2}{M^2} = \frac{du^2}{u^2} \left[1 + \frac{\gamma-1}{2} M^2\right]$, using (10).

Equations (18) and (16) may be combined to yield

$$\frac{dP}{P} = -\left[\frac{1+(\gamma-1)M^2}{2+(\gamma-1)M^2}\right] \frac{dM^2}{M^2} , \qquad (19)$$

and equations (19) and (17) yield

$$\frac{dM^2}{M^2} = \frac{\gamma M^2 \left[1 + \frac{\gamma-1}{2} M^2\right]}{1 - M^2} \frac{fdL}{D} . \qquad (20)$$

The equations derived above are identical in form to those obtained for single phase flow through a constant area duct [53]. However, the form is only cosmetic, since the Mach number has been redefined according to (9) to account for liquid effects.

Equation (20) may be integrated between the duct inlet (station 1) and exit (station 2), to yield

$$\frac{fL}{D} = \frac{1}{\gamma}\left[\frac{M_2^2 - M_1^2}{M_1^2\ M_2^2}\right] + \frac{\gamma+1}{2\gamma}\ \ln\left[\frac{(1+\frac{\gamma-1}{2}M_2^2)\ M_1^2}{(1+\frac{\gamma-1}{2}M_1^2)\ M_2^2}\right]. \qquad (21)$$

Also, (19) integrated between the same two stations yields

$$\frac{P_2}{P_1} = \left[\frac{(2+(\gamma-1)M_1^2)M_1^2}{(2+(\gamma-1)M_2^2)M_2^2}\right]^{1/2}. \qquad (22)$$

Critical flow occurs when the vent exit Mach number, M_2, equals one.

The mass flux per unit area, G, may be written as

$$G = \rho_1 u_1 = M_1\sqrt{\gamma P_1 \rho_1}\ , \qquad (23)$$

using (9) and the continuity equation.

The above expression for G is applicable for the flow between two reservoirs where an area reduction (contraction) is not present, i.e., wall shear represents the only frictional loss. In this case, the flow from station 1 to station 2 is accelerated by the pressure difference $P_{o1} > P_{exit}$.

When an area reduction occurs between the inlet reservoir and the vent, as shown in Figure 1, an isentropic inlet effect is included. In this case, (2) integrated between the reservoir and vent inlet yields

$$XC_p\,(T_{o1} - T_1) = u_1^2/2 \tag{24}$$

where $u_{o1} \approx 0$, for the static reservoir.

Since the flow is adiabatic and homogeneous, the gaseous phase will expand isentropically where

$$Pv_g^{\gamma} = \text{constant},$$

or

$$Pv^{\gamma} = \text{constant}, \tag{25}$$

from (3) for frozen flow. Of course, these constants are not the same. Equations (4) and (25) yield,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}, \tag{26}$$

and thus, (24) becomes

$$\frac{T_{o1}}{T_1} - 1 = \frac{\gamma-1}{2}\,\frac{u_1^2}{\gamma XRT_1} \quad \text{or}$$

$$\frac{P_{o1}}{P_1} = 1 + \frac{\gamma-1}{2}\,M_1^2\;\frac{\gamma}{\gamma-1} \tag{27}$$

The back pressure in the sink node, P_{exit}, is assumed to be the exit static pressure, P_2, where the sink node represents a plenum.

The pressure ratio (22) then becomes,

$$\frac{P_{exit}}{P_{o1}} = \left[\frac{(2+(\gamma-1)M_1^2)M_1^2}{(2+(\gamma-1)M_2^2)M_2^2}\right]^{1/2} \left(1 + \frac{\gamma-1}{2} M_1^2\right)^{\frac{-\gamma}{\gamma-1}} . \qquad (28)$$

For isentropic flow between the reservoir and vent inlet, using the isentropic flow relationships,

$$G = \frac{M_1}{\left(1+ \frac{\gamma-1}{2} M_1^2\right)^{\frac{1}{\gamma-1}}} \left[\frac{\gamma \rho_{o1} P_{o1}}{1+ \frac{\gamma-1}{2} M_1^2}\right]^{1/2} . \qquad (29)$$

Also, several auxiliary equations may be derived to provide additional vent flow data.

Using the expression for dP/P from equation (17), equation (16) becomes

$$\frac{du}{u} = \frac{\gamma M^2}{2(1-M^2)} \frac{fdL}{D} ; \qquad (30)$$

and (17) and (19) yield

$$\frac{dM^2}{M^2} = \frac{[2+(\gamma-1)M^2]\gamma M^2}{2(1-M^2)} \frac{fdL}{D} . \tag{31}$$

Also, equations (30), (31) and (18) imply

$$\frac{dT}{T} = - \frac{\gamma(\gamma-1)M^4}{2(1-M^2)} \frac{fdL}{D} ; \tag{32}$$

and from (30) and (12),

$$\frac{d\rho}{\rho} = - \frac{\gamma M^2}{2(1-M^2)} \frac{fdL}{D} . \tag{33}$$

Next, equation (27) implies

$$\frac{dP_o}{P_o} = \frac{dP}{P} + \frac{\gamma M^2}{2[1+\frac{\gamma-1}{2} M^2]} \frac{dM^2}{M^2} , \tag{34}$$

or from (17) and (31),

$$\frac{dP_o}{P_o} = - \frac{\gamma M^2}{2} \frac{fdL}{D} . \tag{35}$$

All of these equations reduce to the conventional gas dynamics equations [53] for single-phase compressible flow, when the quality is equal to one.

Using (31) to relate the friction factor to the Mach number, (30), (32), (33) and (34) can be integrated to yield:

$$\frac{u_2}{u_1} = \left[\frac{M_2^2}{M_1^2}\left(\frac{2+(\gamma-1)M_1^2}{2+(\gamma-1)M_2^2}\right)\right]^{1/2} = \frac{\rho_1}{\rho_2} , \tag{36}$$

$$\frac{T_2}{T_1} = \frac{2+(\gamma-1)M_1^2}{2+(\gamma-1)M_2^2} , \tag{37}$$

and $$\frac{P_{o2}}{P_{o1}} = \frac{M_1}{M_2}\left[\frac{2+(\gamma-1)M_2^2}{2+(\gamma-1)M_1^2}\right]^{\frac{\gamma+1}{2(\gamma-1)}} . \tag{38}$$

Equation (38) is included only for the sake of completeness; (28) is used for the total pressure ratio for the actual vent flow.

The sonic conditions are defined to occur when the exit Mach number reaches one. The critical pressure ratio becomes

$$\left(\frac{P_{exit}}{P_{o1}}\right)^* = \left[\frac{M_1^2}{1+\frac{\gamma-1}{2}}\right]^{1/2} \left[1 + \frac{\gamma-1}{2} M_1^2\right]^{\frac{1+\gamma}{2(1-\gamma)}} . \tag{39}$$

When the given pressure ratio, P_{exit}/P_{o1} is less than the critical pressure ratio, the flow is taken to be critical.

III. SOLUTION TECHNIQUE AND RESULTS

Results are obtaned by the simultaneous solution of equations (21) and (22) or (28) for M_1 and M_2. Equation (22) is used when inlet effects are ignored, e.g., there is no connecting duct between stations one and two. In this case, the stagnation pressures are used in place of the static pressures in equation (22), and the flow is artificially accelerated between stations one and two. Equation (28) is used when inlet effects are included, or a flow restriction (or duct) exists between stations one and two. The total pressure in the nodes upstream and downstream of the vent is assumed known. The friction factor is given or can be expressed as a function of the flow field parameters. This functional form of the friction factor can then be expressed in terms of the inlet and exit Mach numbers using the relationships given previously. The governing equations are then more complicated, but can still be solved simultaneously for M_1 and M_2. The two governing equations are reduced to one nonlinear algebraic equation. The critical flow equations are solved for M_2 equal one and the resulting critical pressure ratio is calculated. This critical pressure ratio is compared to the actual pressure ratio to determine if the flow is choked. If the flow is choked, the mass flux is calculated. If the flow is subsonic, the governing equations are solved simultaneously for M_1 and M_2, and then the mass flux is calculated. Once the inlet and exit Mach numbers are known, other flow field parameters can be readily calculated.

The actual solution is achieved by attempting to solve the final nonlinear equations using the Newton-Raphson method. If this method does not converge, solve the equations by the Bisection method.

Several parametric studies utilizing this flow model have been performed and inconsistent behavior was not observed. Figures 2A, 2B, 2C and 2D present the mass flow rate versus inlet pressure for a specified back pressure. The quality and friction factor were varied. A steam-water mixture was considered with $\gamma=1.1$, although, in general, γ is calculated from the known reservior data. Figures 2A and 2B include the isentropic inlet effect; Figures 2C and 2D do not.

Figures 3A, 3B and 3C compare the critical flow rates versus critical pressure to the analytical data of Fauske [8] for steam-water mixtures. Two of Fauske's models are shown [8]. Comparison is also made to the experimental data of Faletti, Moy, Fauske [8], also [4], [10]; and Klingebiel [13]. Figures 3 indicate good agreement with the experimental data and the theoretical predictions of Fauske, especially at high qualities ($X>.5$). The results of this model also agree very well with Moody's theoretical results as presented in [47]. For a quality of one, the equations associated with the vent flow model reduce to the classical gas dynamics expressions [53], as expected. Thus,

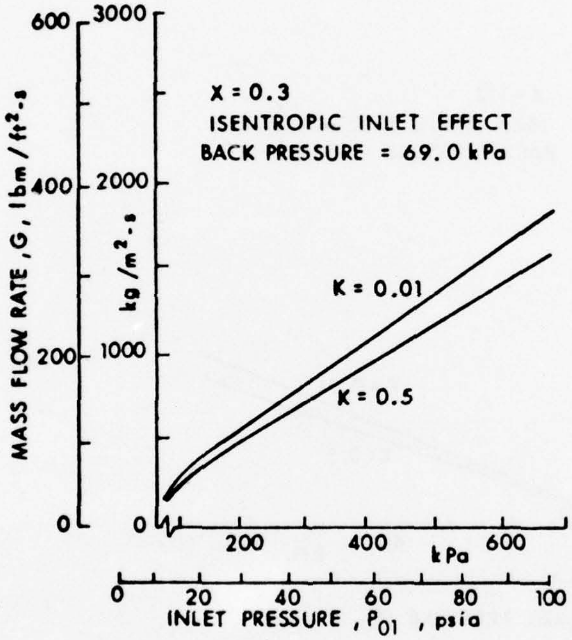

Figure 2A. Vent Inlet Pressure Versus Mass Flow Rate

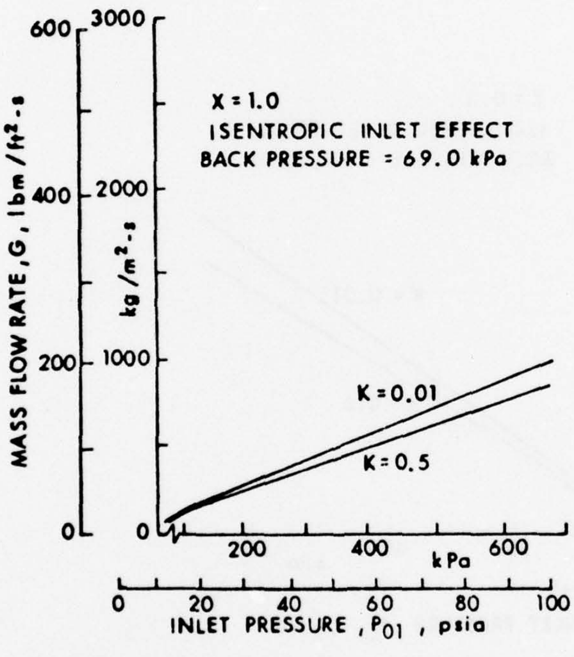

Figure 2B. Vent Inlet Pressure Versus Mass Flow Rate

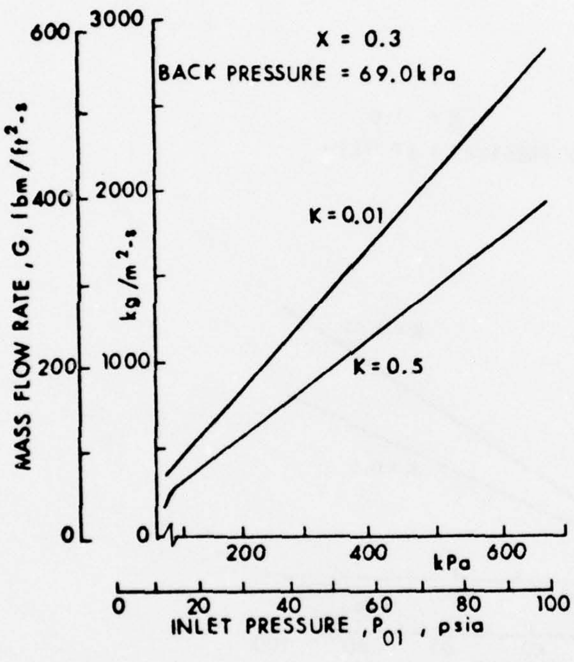

Figure 2C. Vent Inlet Pressure Versus Mass Flow Rate

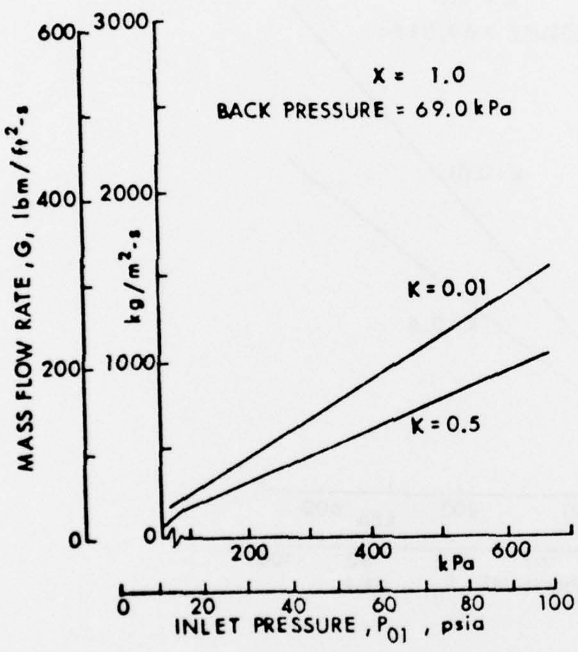

Figure 2D. Vent Inlet Pressure Versus Mass Flow Rate

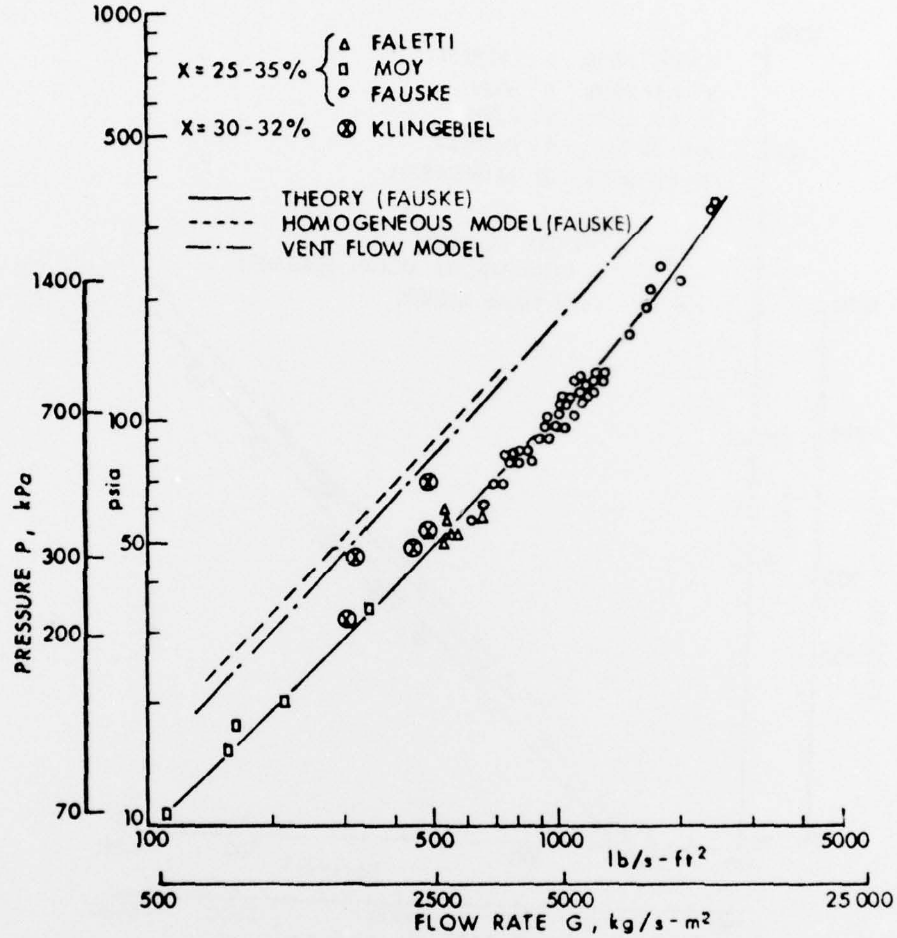

Figure 3A. Comparison with Data from [8] and [13], X = 30%

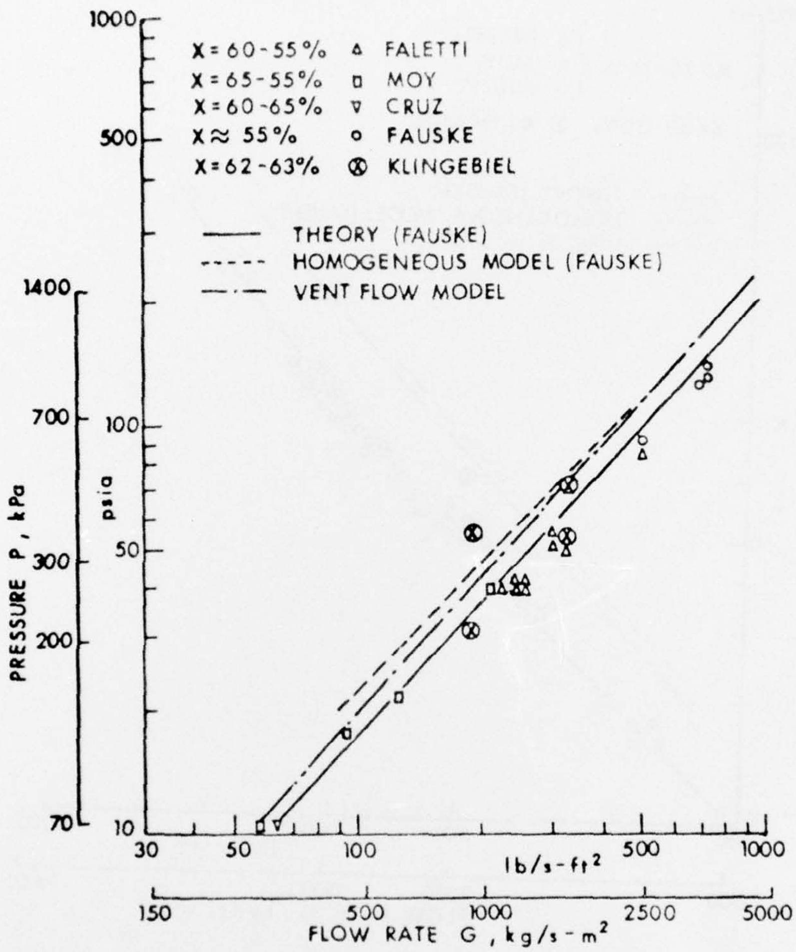

Figure 3B. Comparison with Data from [8] and [13], Continued, X = 60%

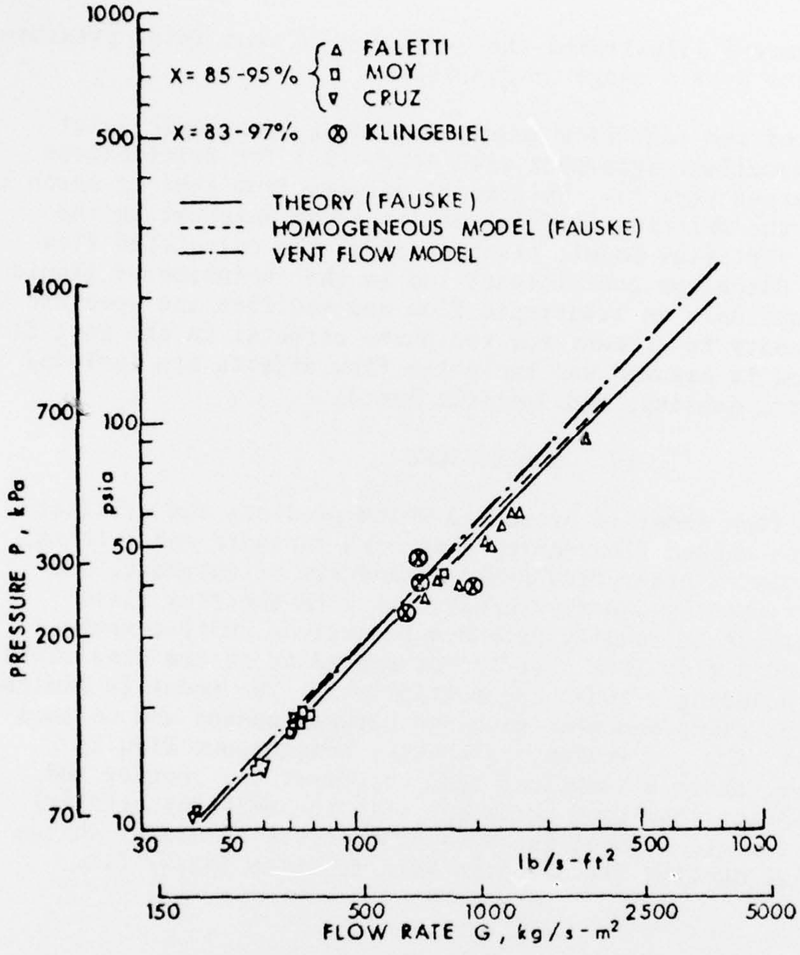

Figure 3C. Comparison with Data from [8] and [13], Continued, X = 90%

the theory is exact for a quality of one; excellent for qualities above 0.5; acceptable for qualities as low as 0.2; and poor for qualities less than 0.1.

Finally, Figure 4 illustrates the variation of vent inlet pressure with mass flow rate over a range of qualities.

The results of the vent flow model, including isentropic inlet effects exhibit excellent agreement with Aisch [22] for frictionless flow using a constant $\gamma(=1.1)$. This model differs from that of Aisch in the treatment of the friction coefficient (it is an inherent in the equations of the vent flow model; Aisch modifies the calculated flow rate by use of a discharge coefficient) and in the inclusion of liquid effects (Aisch considers an isentropic flow and modifies the specific heat ratio and the density to account for two-phase effects; in the vent flow model, frozen flow is assumed and two-phase flow effects are included in the sonic velocity, density, and specific heats).

IV. CONCLUSIONS

A two-phase flow model is presented which predicts the critical pressure ratio for choked flow, calculates both subsonic and critical flow rates, calculates other flow field parameters of interest, and utilizes a sonic velocity equation consistent with the flow field model. The model can be readily extended to include multi-component two-phase flow and a flow loss coefficient dependent on the flow field parameters (or including a two-phase multiplier). The model is limited by the assumptions of no momentum exchange between phases and no mass or heat transport, i.e., a frozen, adiabatic, homogeneous flow is assumed. However, these assumptions tend to offset one another and the predicted results show good agreement with the existing critical flow data. The vent flow model is capable of providing useful design analyses in a high quality ($X>.3$), high void fraction ($\alpha>.9$) flow regime.

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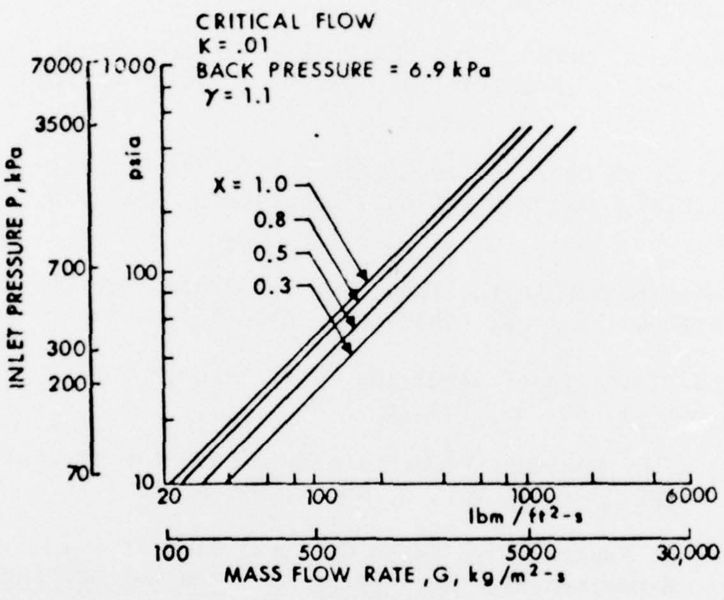

Figure 4. Vent Inlet Pressure Versus Mass Flow Rate

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LIST OF SYMBOLS

a	Sonic Velocity
α	Void Fraction
C_p	Coefficient of Specific Heat at Constant Pressure
dL	Increment of Vent Length
D	Vent Hydraulic Diameter
f	Resistance Coefficient
$\frac{fL}{D} = K$	Friction Factor
G	Mass Flow Rate
h	Enthalpy
γ	Specific Heat Ratio
L	Vent Length
M	Mach Number
m	Mass
P	Pressure
R	Gas Constant
T	Temperature
ρ	Density
u	Velocity
v	Specific Volume
x	Quality

LIST OF SYMBOLS (Cont.)

Subscripts

g	Vapor
f	Liquid
1	Vent inlet
2	Vent exit
0	Stagnation or Reservoir
TP	Two-Phase

Superscripts

*	Sonic or Critical Condition

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